

# Spacecraft Radiator Freeze Protection Using a Regenerative Heat Exchanger with Bypass Setpoint Temperature Control

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## ***Abstract***

Spacecraft radiators are sized for their maximum heat load in their warmest thermal environment, but must operate at reduced heat loads and in colder environments. For systems where the radiator environment can be colder than the working fluid freezing temperature, radiator freezing becomes an issue. Radiator freezing has not been a major issue for the Space Shuttle and the International Space Station (ISS) active thermal control systems (ATCSs) because they operate in environments that are warm relative to the freezing point of their external coolants (Freon-21 and ammonia, respectively).

For a vehicle that lands at the Lunar South Pole, the design thermal environment is 215K, but the radiator working fluid must also be kept from freezing during the 0 K sink of transit. A radiator bypass flow control design such as those used on the Space Shuttle and ISS requires more than 30% of the design heat load to avoid radiator freezing during transit - even with a very low freezing point working fluid. By changing the traditional ATCS architecture to include a regenerating heat exchanger inboard of the radiator and by using a regenerator bypass flow control valve to maintain system setpoint, the required minimum heat load can be reduced by more than half. This gives the spacecraft much more flexibility in design and operation.

The present work describes the regenerator bypass ATCS setpoint control methodology. It includes analytical results comparing the performance of this system to the traditional radiator bypass system. Finally, a summary of the advantages of the regenerator bypass system are presented.

## ***Background***

Active thermal control systems (ATCSs) on human spacecraft provide fluid at a controlled temperature to cool equipment, cool the crew cabin air, and provide condensation control. When spacecraft radiators are the major source of heat

rejection, as was the case on the Apollo Service Module, is the case on the Space Shuttle Orbiter and the International Space Station (ISS), and will be the case on the upcoming Crew Exploration Vehicle (CEV), their coolant return temperature is normally set by controlling the radiator bypass flow. That is, a preset coolant return temperature is maintained by mixing a variable amount of warm coolant from the radiator inlet with the cold radiator outlet flow. All four vehicles listed above use this approach.

Spacecraft radiators are sized for their maximum heat load in their warmest thermal environment, but must operate at reduced heat loads and in colder environments. For systems where the radiator environment can be colder than the working fluid freezing temperature, radiator freezing becomes an issue. Radiator freezing has not been a major issue for the Space Shuttle and ISS ATCSs because they operate in environments that are warm relative to the freezing point of their coolants (Freon-21 and ammonia, respectively). The Apollo Service Module flew through extremely cold environments during its trip to the Moon. The minimum ATCS heat load required to avoid freezing was reduced to manageable values by use of a stagnation radiator - where the radiator fin efficiency is greatly reduced at low heat loads by stagnating the flow through much of the radiator.

For a vehicle that lands at the Lunar South Pole, the design thermal environment is 215K<sup>1</sup>. The radiator working fluid must also be kept from freezing during the 0 K sink of transit. Because only safe working fluids can now be used in the crew cabin, an Apollo-like single internal/external ethylene glycol/water loop is not an option. If the allowed propylene glycol/water mixture is used in a stagnation radiator, its higher freezing point results in an allowable heat load range greatly reduced from Apollo's range of 65% to 100% of full load. At best, such a design would have a very narrow operating range. At worst, such a design is not feasible.

Even with a low freezing point working fluid in a traditional radiator bypass flow control design, a large fraction of the design heat load would be required to avoid radiator freezing during transit to the Moon. An alternate ATCS architecture was developed to provide a flexible system for transit to and operation on the Moon.

## ***System Design and Operation***

### **Radiator Bypass Setpoint Control ATCS**

Overview - A simplified radiator bypass thermal system is shown in Figure 1. The standard two loop design includes an internal loop in the pressurized volume

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<sup>1</sup> Assuming that the ground slopes away from the vehicle at 5° and adding in the 1.54° tilt of the lunar axis gives a worst case sun angle of 6.54°. The lunar surface optics are  $\alpha/\epsilon=0.865/0.95$ , so a simple energy balance using a deep space (0 K) view and a solar constant of 1371 W/m<sup>2</sup> yields a Lunar surface temperature of 224 K. A vertical cylindrical radiator with a 10 mil silver-Teflon coating ( $\alpha/\epsilon=0.10/0.85$ ) has an average sink temperature of 215 K

plus an external loop. This way the internal loop can use a non-toxic fluid with good heat transfer characteristics (like water) and the external loop can use a low freezing point fluid that may be toxic. The two loops are connected by an interloop heat exchanger. The loop flow rates are matched so that the product of mass flow rate and specific heat is the same in the two loops, maximizing the average radiator temperature and heat rejection capability. A radiator bypass controls the mixed radiator fluid return temperature. A similar design is used in the Space Shuttle Orbiter and in the international Space Station (ISS) ATCS. The radiator performance in this type of system can be explored using a simple analytical model.

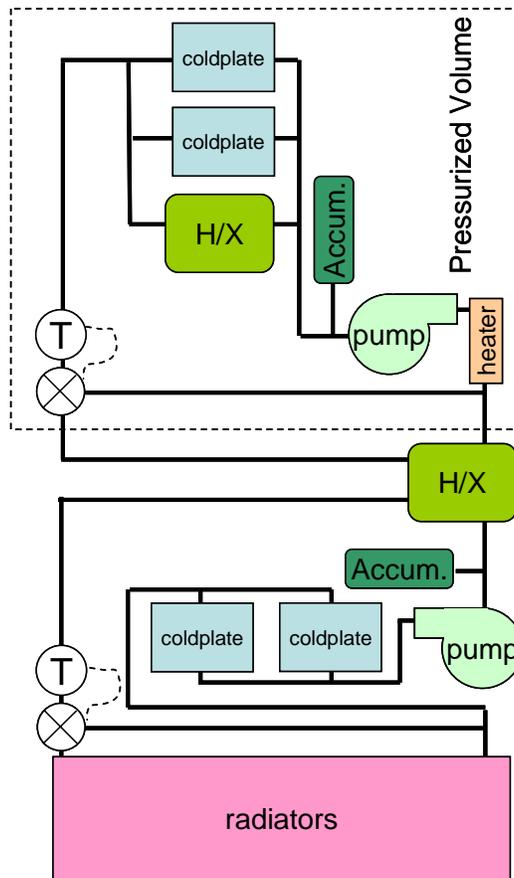


Figure 1 –Active Thermal Control Design with Radiator Bypass

Radiator Analysis - The performance of the radiator bypass ATCS was explored using a simplified Excel thermal model. The model predicts the radiator flow and temperature distribution using a number of simplifying assumptions.

The differential local heat rejection,  $dQ$ , from a given differential area of radiator,  $dA$ , is

$$dQ = \epsilon \eta \sigma dA (T_{\text{root}}^4 - T_{\infty}^4) \quad (1)$$

where  $\varepsilon$  is the radiator emissivity,  $\eta$  is the radiator fin efficiency,  $\sigma$  is the Stefan-Boltzman constant,  $T_{\text{root}}$  is the radiator root temperature, and  $T_{\infty}$  is the radiator environment temperature.

If the liquid flow tubes are evenly spaced in the radiator and have uniform flow, the differential temperature drop for the radiator fluid,  $dT$ , is

$$dT = \frac{dQ}{\dot{m}c_p} \quad (2)$$

where  $\dot{m}$  is the radiator mass flow rate and  $c_p$  is the radiator fluid specific heat.

If

- the fin root temperature is identical to the local fluid temperature
- total radiator area is expressed as the product of its length,  $L$ , and width,  $w$
- $\zeta$  is defined as  $x/L$ , the non-dimensionalized length along the flow path

equations (1) and (2) can be combined as

$$\dot{m}c_p dT = \varepsilon\eta\sigma wLd\zeta(T^4 - T_{\infty}^4) \quad (3)$$

or

$$\frac{dT}{d\zeta} = \frac{\varepsilon\eta\sigma A}{\dot{m}c_p}(T^4 - T_{\infty}^4) \quad (4)$$

By representing the radiator mass flow rate as a fraction,  $F_{\text{radiator}}$ , of the full mass flow rate,  $\dot{m}_{\text{max}}$ , eqn. (4) becomes

$$\frac{dT}{d\zeta} = \frac{\varepsilon\eta\sigma A}{F_{\text{radiator}}\dot{m}_{\text{max}}c_p}(T^4 - T_{\infty}^4) \quad (5)$$

Making the further simplifications that

- the radiator fin efficiency is constant
- the fluid specific heat is constant

allows all the constants in eqn. (5) to be represented by a single constant,  $C$

$$\frac{dT}{d\zeta} = \frac{C}{F_{\text{radiator}}}(T^4 - T_{\infty}^4) \quad (6)$$

This equation allows the heat transfer to be tracked through the radiator by following only the fluid temperature.

For the ATCS modeled here, the key temperatures are:

- The setpoint temperature,  $T_{\text{setpoint}}$ , is 283 K
  - this is the mixed outlet temperature at all times and is the radiator outlet temperature at full load in the design environment

- The external loop temperature drop at the design point,  $\Delta T_{\text{design}}$ , is 20 K
  - this is the temperature difference between the radiator inlet temperature and the setpoint temperature at all times
  - notably, this is also the temperature rise across the internal loop at full load
- The design sink temperature is 215 K

The radiator performance is analyzed by calculating C for the design case (by following the radiator fluid through the radiator). Once C is found, the radiator can now be analyzed to calculate the radiator flow fraction,  $F_{\text{radiator}}$ , that yields the desired radiator mix temperature at any heat load and in any environment. The setpoint is maintained when the following equation is satisfied

$$T_{\text{setpoint}} = (1 - F_{\text{radiator}})T_{\text{radiator,in}} + F_{\text{radiator}} T_{\text{radiator,out}} \quad (7)$$

where  $T_{\text{radiator,out}}$  is the calculated radiator outlet temperature. The radiator inlet temperature,  $T_{\text{radiator,in}}$  is calculated from an energy balance

$$T_{\text{radiator,in}} = T_{\text{setpoint}} + \Delta T_{\text{design}} F_{\text{load}} \quad (8)$$

where  $F_{\text{load}}$  is the fraction of full load.

The radiator performance in the 215 K design sink is shown in Figure 2. The figure shows that the radiator flow decreases non-linearly with heat load. The flow fraction at 50% of peak load is less than 20%.

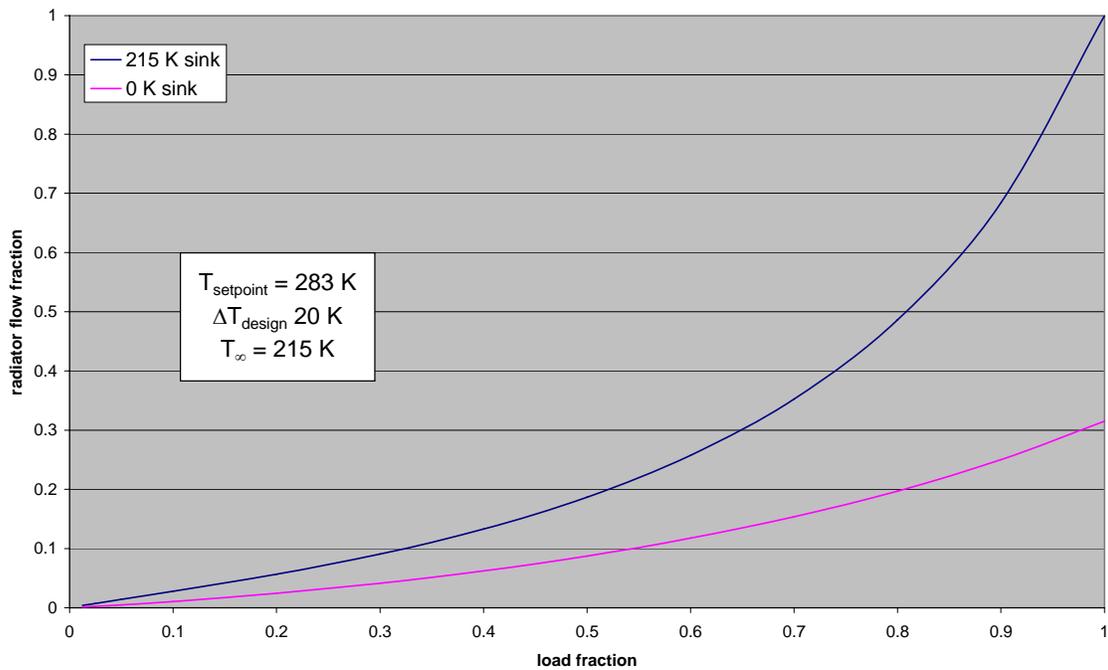


Figure 2 – Radiator Bypass System Radiator Flow

Figure 2 also shows the change in radiator flow fraction with heat load in a 0 K sink. At design heat load, the radiator flow is approximately 30% and it drops off non-linearly as the heat load is decreased.

Figure 3 shows the radiator outlet temperature as a function of heat load fraction for the two heat sinks. The freezing points of several heat transfer fluids are also noted on the graph. The figure shows that an ammonia external loop requires more than 60% of the design heat load to avoid radiator freezing in a deep space environment. Even with Freon-21<sup>2</sup>, which is a less desirable working fluid owing to its poor heat transfer properties, the heat load must be maintained at more than 25% of the design heat load to avoid freezing. The freezing point of HFC-245fa is also noted in the graph. It is a “green” analog of the Freon-21 that is used in the Space Shuttle Orbiter external ATCS and has a lower freezing point than ammonia.

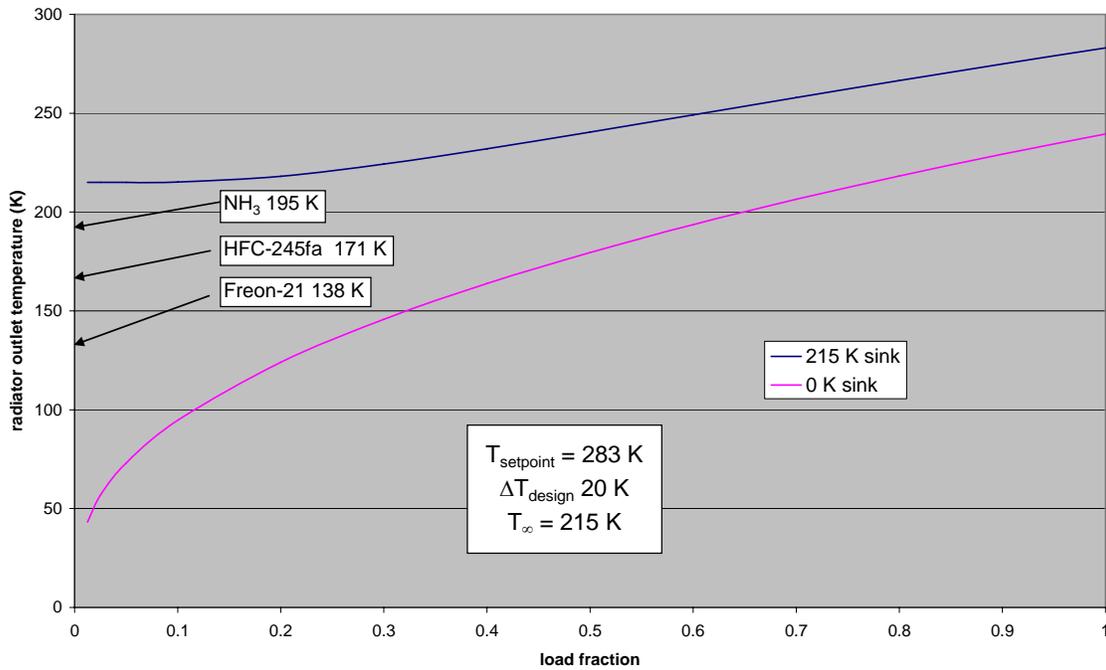


Figure 3 – Radiator Bypass System Radiator Outlet Temperature

The minimum required heat load required in a radiator bypass system is much larger than the heat rejection if the entire radiator were at the fluid freezing point (the absolute minimum required heat load) because:

- the cold environment causes a cold radiator return temperature.

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<sup>2</sup> Freon-21 is no longer manufactured owing to its high ozone depletion potential; however NASA has stockpiles for use on the Space Shuttle Orbiter ATCS that could be made available for future spacecraft.

- cold radiator return temperature causes the radiator bypass valve to open, mixing warm radiator inlet flow with the return flow to maintain the required mix temperature.
- the resulting decreased radiator flow further decreases the radiator return temperature which requires the bypass valve to open further.

This positive feedback loop results in very low flows through the radiator in cold environments. In fact, at the freeze limit only a small fraction of the total loop flow is passing through the radiator. These low flows require relatively high heat loads to avoid freezing.

### **Regenerator Bypass Setpoint Control ATCS**

Overview - To address the issue of high required minimum system heat load during deep space transit, a novel setpoint temperature control methodology was conceived. Rather than using a radiator bypass valve, a regenerative heat exchanger is placed inboard of the radiators. A cold side bypass valve is used to set the return temperature. The resulting loop architecture is shown in Figure 4.

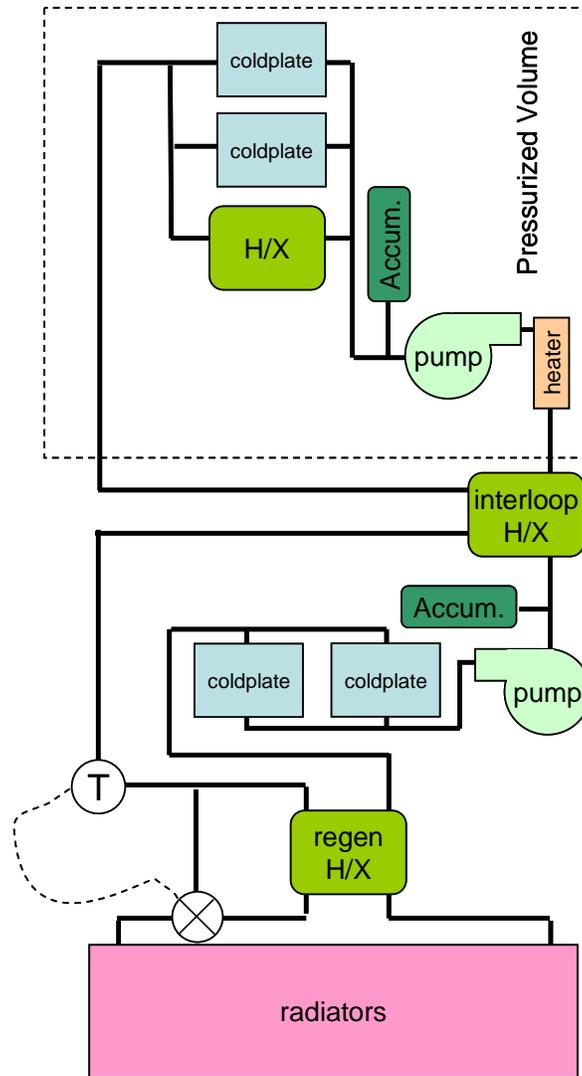


Figure 4 –Active Thermal Control Design with Regenerator Bypass

During operation, the regenerator bypass flow is varied, mixing cold radiator return fluid and warm regenerator outlet fluid to maintain the system setpoint. The regenerator is fully bypassed at the highest heat load in the design environment, sending all the flow to the radiator - just as in the traditional two loop system. At the lowest low heat load for stable operation, the bypass flow is closed off – sending all the flow through the regenerator. This lowers the radiator inlet temperature well below the system setpoint while maintaining full flow through the radiators. By dramatically decreasing the average radiator temperature at low heat loads, this concept substantially reduces the heat load required to avoid freezing.

System Analysis - The radiator model described in the previous section was modified to explore the performance of the Regenerator Bypass ATCS. The model was modified to include the regenerator and energy balances were included across key components.

The key temperatures are identical to the traditional two loop analysis:

- The setpoint temperature is 283 K
  - here it is the regenerator mix temperature
- The external loop temperature drop is 20 K at the design point
  - maximum load in design sink temperature – full regenerator bypass
- The design sink temperature is 215 K

Because the same design conditions are used as for the two loop system, the value of C is also the same.

Off-design cases were analyzed for the minimum required heat load by assuming that the regenerator bypass was closed. The radiator flow fraction was unity and the required radiator inlet temperature was calculated for a given outlet temperature using eqn (6). This yielded the heat load fraction,  $F_{load,min}$ , from a simple ratio

$$F_{load,min} = \frac{T_{radiator,in} - T_{radiator,out}}{\Delta T_{design}} \quad (9)$$

Figure 5 shows the minimum radiator temperature as a function of heat load. . The figure shows that heat loads as low as 1/3<sup>rd</sup> are possible for an ammonia system operating in a deep space sink. If Freon-21 is used as the working fluid, heat loads less than 10% of full load are achievable.

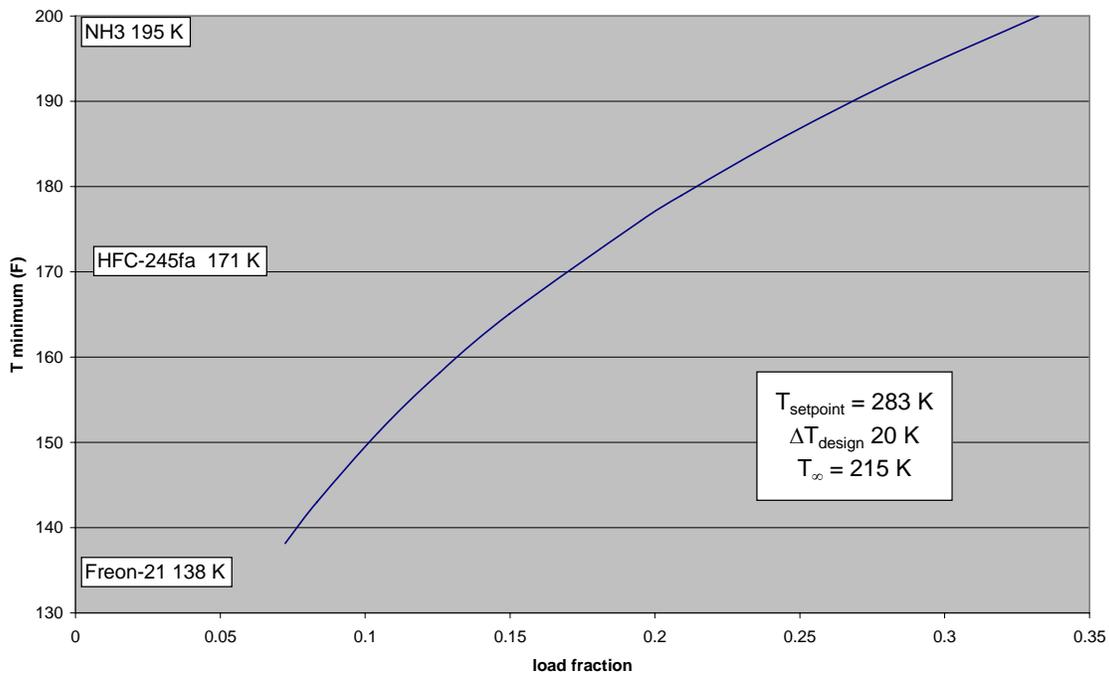


Figure 5 – Highest Minimum Radiator Temperature at Various Heat Loads  
Regenerator Bypass System –  $T_{sink} = 0 \text{ K}$

The required regenerator performance can be calculated from its heat load and internal temperature difference. The required design heat load,  $Q_{\text{regen,design}}$ , is

$$Q_{\text{regen,design}} = \dot{m}_{\text{max}} c_p (T_{\text{setpoint}} - T_{\text{radiator,out}}) \quad (10)$$

The loop design heat load,  $Q_{\text{loop,design}}$ , is defined as

$$Q_{\text{loop,design}} = \dot{m}_{\text{max}} c_p \Delta T_{\text{design}} \quad (11)$$

so

$$Q_{\text{regen,design}} = Q_{\text{loop,design}} \frac{(T_{\text{setpoint}} - T_{\text{radiator,out}})}{\Delta T_{\text{design}}} \quad (12)$$

Since  $(T_{\text{setpoint}} - T_{\text{radiator,out}})/\Delta T_{\text{design}}$  is greater than unity, the regenerator heat transfer will always be a multiple of the system design heat load.

The regenerator internal temperature difference at zero bypass is  $T_{\text{radiator,in}} - T_{\text{radiator,out}}$ , so, using eqns. (9) and (10), the required regenerator performance,  $UA_{\text{regen,design}}$ , is

$$UA_{\text{regen,design}} = \frac{\dot{m} c_p (T_{\text{setpoint}} - T_{\text{radiator,out}})}{F_{\text{load}} \Delta T_{\text{design}}} \quad (13)$$

or, including eqn. (11)

$$\frac{UA_{\text{regen,design}}}{Q_{\text{loop,design}}} = \frac{(T_{\text{set}} - T_{\text{radiator,out}})}{F_{\text{load}} \Delta T_{\text{design}}^2} \quad (14)$$

Figure 6 shows the regenerator performance required to achieve the desired minimum radiator temperatures. The regenerator performance is given as  $UA_{\text{regenerator}}/Q_{\text{loop,design}}$ . As the desired radiator outlet temperature decreases, the minimum heat load fraction decreases (as shown in Figure 6). This decreases the driving temperature difference across the heat exchanger. These two factors work together geometrically to increase the required regenerator performance at lower minimum heat loads.

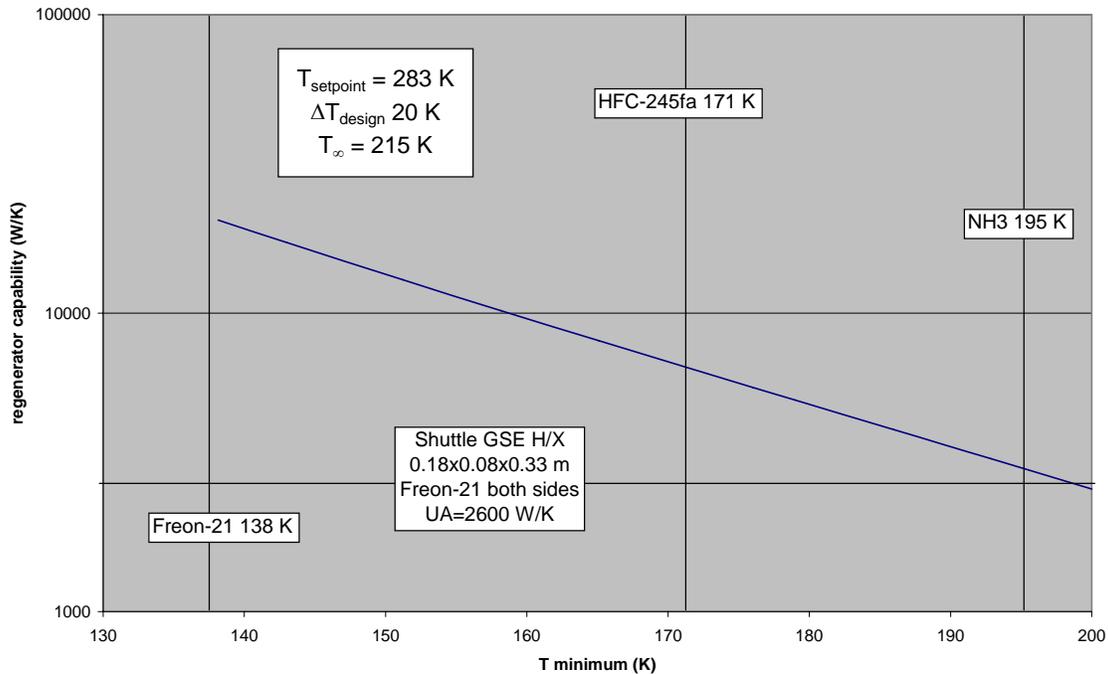


Figure 6 – Regenerator Sizing  
Regenerator Bypass System –  $T_{\text{sink}} = 0 \text{ K}$

Figure 6 shows that, for a 3 kW ammonia external system, a regenerator similar in size to the Space Shuttle ground support heat exchanger (GSE H/X) is sufficient – even before accounting for the superior heat transfer properties of ammonia vs. Freon-21.

### **System Comparison**

Table 1 shows a comparison of the two two-loop systems. The minimum allowable temperatures are taken as the working fluid freezing point plus a 5 K margin. The table shows that the regenerator bypass system yields a much larger performance range for a given set of system design specifications. It shows that, for each of the commonly used heat transfer fluids, the regenerator bypass system can operate at less than half of the minimum heat load of the traditional radiator bypass system. On a case by case basis, the additional mass of the regenerator may be worthwhile for the design flexibility that it offers. The table also shows that an ammonia regenerator system has a comparable turndown to a traditional Freon-21 system. With similar requirements, the superior performance of ammonia as a heat transfer fluid will likely outweigh the additional mass of a regenerator.

	Traditional Two-Loop min load fraction	Regenerator Bypass System min. load fraction	Regenerator Size $\frac{UA_{\text{regenerator}}}{Q_{\text{design}}}$ (1/K)
<u>Ammonia</u> $T_{\text{freeze}} = 195 \text{ K}$ $T_{\text{minimum}} = 200 \text{ K}$	0.65	0.33	650
<u>HFC-245fa</u> $T_{\text{freeze}} = 171 \text{ K}$ $T_{\text{minimum}} = 176 \text{ K}$	0.48	0.20	1400
<u>Freon-21</u> $T_{\text{freeze}} = 138 \text{ K}$ $T_{\text{minimum}} = 143 \text{ K}$	0.29	0.08	5100

Table 1 – Loop Performance Comparison

### ***Conclusion***

A new methodology for spacecraft active thermal control system setpoint control has been described and analyzed. The regenerator bypass system offers a greatly expanded allowable system heat load turndown for a given working fluid by replacing the radiator bypass with a regenerator and bypass. It also allows the substitution of higher performance working fluids for the same design parameters. This ATCS architecture holds promise for decreasing system mass and for increasing operational flexibility.